High Compression Turbopumps: Functional Principle and Application

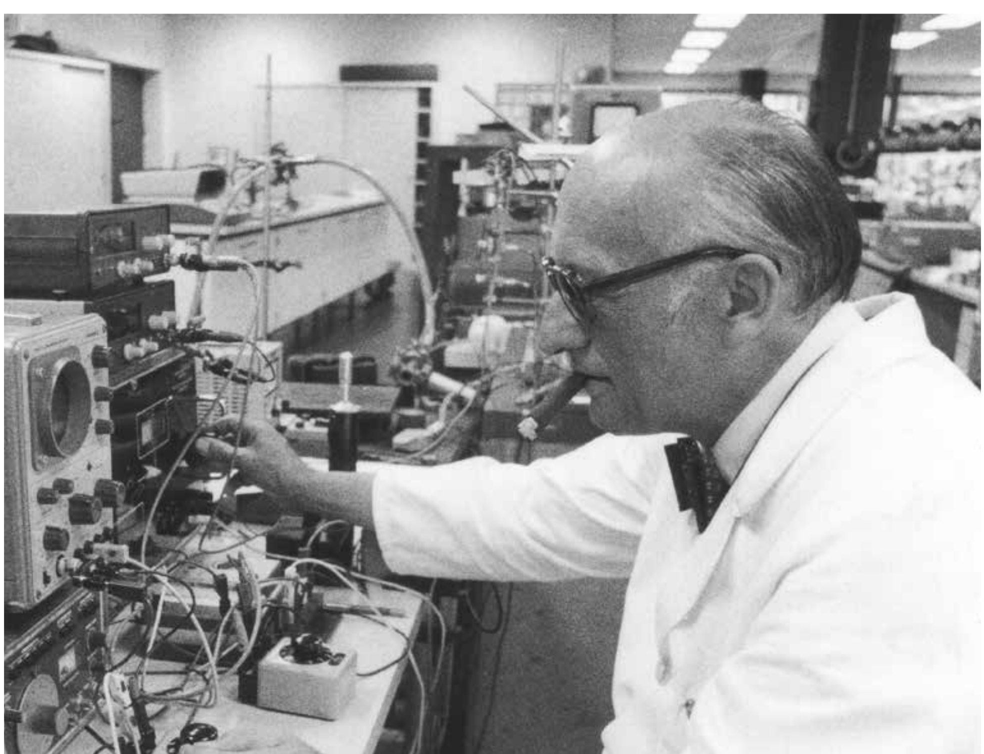
A major milestone in Pfeiffer Vacuum's history was the invention of the turbopump in 1955. Since then, some 3,400 of the company's employees have been working to continually improve vacuum technology. Most recently, this was achieved through the invention of laser balancing technology. This ensures an even longer service life and significantly lower vibrations and noise emissions. But let's start from the beginning: What is the basis for the functional principle of the classic turbopump? And how do you select the right (backing) pump for your application?

PFEIFFER VACUUM

The invention of the turbopump

The first turbopump was invented in 1955. At that time, Dr. Willi Becker had been the head of the technical laboratory at Arthur Pfeiffer Vakuumtechnik GmbH (now Pfeiffer Vacuum) for 13 years. He was concerned with the question of how to prevent the oil from flowing back into the pump housing in oil diffusion pumps. For this purpose, he used a baffle in the form of a rotating fan wheel. Using this setup, gas particles flowed in the direction of the pressure gradient without significant conductance losses. In the opposite direction, backflowing oil molecules were reflected by the rotating fan wheel. This prevented the molecules from reaching the high vacuum side.

In further research, Dr. Becker noticed that this setup not only reduced the oil backflow from the diffusion pump, it also produced a lower total pressure. He then applied a rotor-stator combination and multiple pump stages in series. For his setup, he used the double-flow version – a rotor that was belt-driven to reach a speed of 16,000 rpm. Weighing 62 kg and with a pumping speed of 900 m^3/h, the pump was patented in 1956 and was the forerunner of all today's turbopumps. In 1958, it was presented for the first time at the International Vacuum Congress in Namur, Belgium. Without this invention, our modern life would be unthinkable – because without turbopumps, many manufacturing steps for the production of semiconductors as well as countless coating processes would not be possible.



Dr. Willi Becker, 1958 in the laboratory of Arthur Pfeiffer Vakuumtechnik GmbH (today Pfeiffer Vacuum)

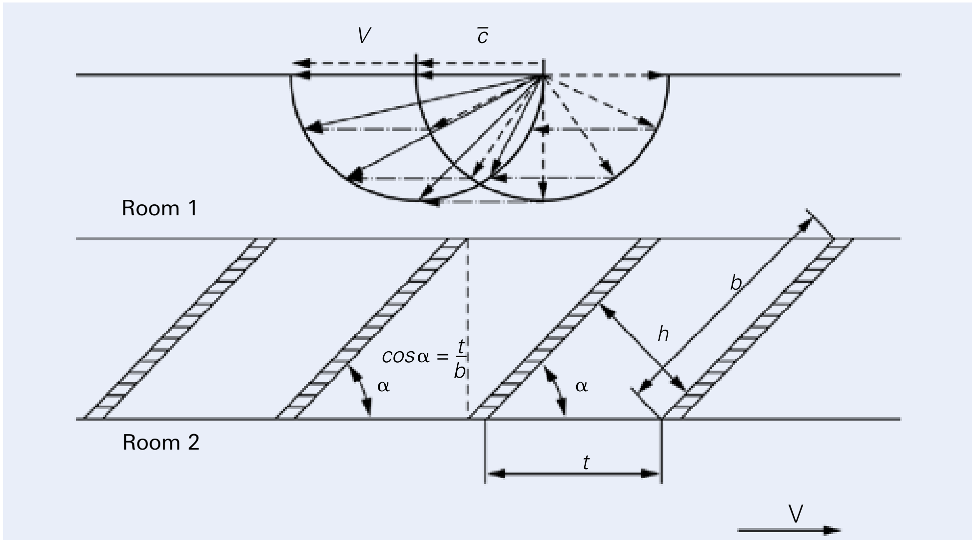


Figure 1: Functional principle of the turbopump

Functional principle and compression ratio

How does a turbopump work? The transfer of momentum from the rapidly rotating blades to the gas molecules to be pumped is the cornerstone of the pumping action of the arrangement of rotor and stator blades (Figure 1). Molecules hitting the blades are adsorbed there and leave the blade again after a short time. The vane velocity v adds up to the thermal molecular velocity c. The thermal molecular velocity c is the speed at which the molecules leave the pump. Molecular flow must prevail in the pump. Otherwise, the velocity component transferred by the blade would be lost through collisions with other molecules. The mean free path T must therefore be greater than the channel height h. During the pumping of gas, backpressure occurs in kinetic pumps, causing backflow. S_0 denotes the pumping speed without backpressure. It decreases with increasing backpressure and reaches the value 0 at the maximum compression ratio *K.*


Figure 2: Arrangement of rotor and stator blades

The compression ratio K_0 can be estimated according to Gaede [1]. For the optically dense blade structure (Figure 1), Gaede's formula applies:

$$K_0 = \frac{p_V}{p_A} = \exp\left(\frac{vL}{\bar{c} \cdot g \cdot h}\right)$$

Gaede's formula

Where:

p_V = Fore-vacuum pressure
p_A = Suction pressure
v = Blade velocity
$\bar{c}$ = average thermal molecular velocity
L = channel length
h = Channel height
g = Factor for specifying the mean impact distance in multiples of the channel height (with $1 < g < 3$)

Replacing the formula v by $v - \cos\alpha$, L by b and h by $t - \sin\alpha$ in the graph, we obtain

$$K_0 = \exp\left(\frac{v \cdot b}{\bar{c} \cdot g \cdot t \cdot \tan\alpha}\right).$$

According to Gaede's estimation, the blades are assumed to be optically dense
and thus satisfy the condition $\cos\alpha = t/b$ (see Figure 1). For larger blade spacing, this implies decreasing compression:

$$K_0 = \exp\left(\frac{1}{g} \cdot \frac{1}{\sin\alpha} \cdot \frac{v}{\bar{c}}\right)$$

The geometrical ratios are taken from figure 1. The factor g is between 1 and 3 [2]. K_0 thus increases exponentially with the blade velocity v and $\sqrt{M}$ aan,

$$\bar{c} = \sqrt{\frac{8 \cdot R \cdot T}{\pi \cdot M}}.$$

R is the universal gas constant,
T is the thermodynamic temperature and
M is the molecular mass.

The compression ratio for nitrogen, for example, is therefore much higher than that for hydrogen.

Calculation of pumping speed

The pumping speed S_0 is proportional to the suction area A and the mean circulation speed of the blades v, i.e. the rotational speed. If the blade angle α is taken into account, this is obtained:

$$S_0 = \frac{1}{2} \cdot A \cdot v \cdot \sin\alpha \cdot \cos\alpha = \frac{1}{4} \cdot A \cdot v \cdot \sin 2\alpha$$

Taking into account the inlet conductance of the flange

$$L_F = \frac{\bar{c}}{4} \cdot A$$

and the optimum blade angle of 45°, the effective pumping speed Seff of a turbopump for heavy gases (molecular weight > 20) is approximated by the formula

$$S_{eff} = \frac{S_0 + L_F}{S_0 \cdot L_F} = \frac{A \cdot v}{4 \cdot \left(\frac{v}{\bar{c}} + 1\right)}$$

By dividing the effective pumping speed by the bladed inlet area of the uppermost disk and taking into account the area obstructed by the blade thickness with the aid of the factor df ≈ 0.9, the maximum specific pumping speed SA of a turbopump is obtained (plotted in figure 2 as a curve for nitrogen as an example gas):

$$S_A = \frac{S_{eff}}{A} = d_f \cdot \frac{v}{4} \cdot \left(\frac{v}{\bar{c}} + 1\right)^{-1}$$

The specific pumping speed in $l \cdot s^{-1} \cdot cm^{-2}$ is plotted on the Y-axis of figure 3, and the mean blade velocity $v = \pi \cdot f \cdot (R_a + R_i)$ is plotted on the X-axis for the circulating frequency f and the outer(R_a) and inner(R_i) radius of the blades. Moving vertically upward from a selected point on the X-axis, the intersection with the curve shows the maximum specific pumping speed of the pump S_A for that speed. Multiplying by the bladed area of the input disk: $A = (R_a^2 - R_i^2) \cdot \pi$, gives the pumping speed.

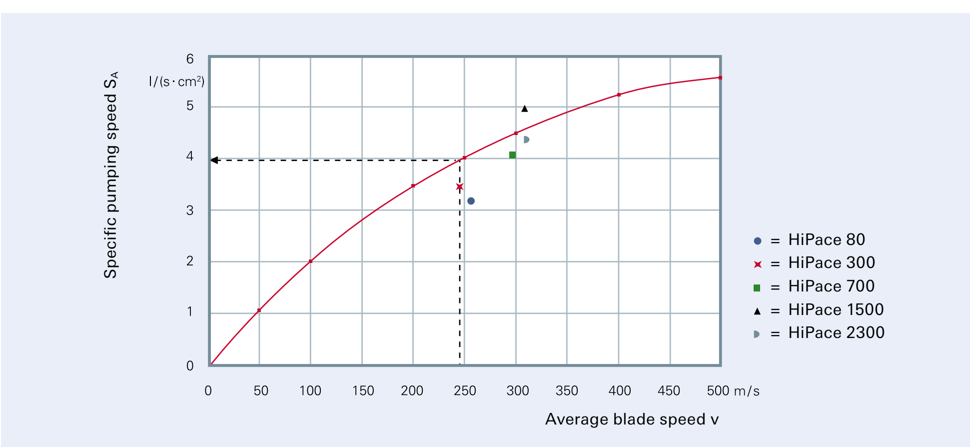


Figure 3: Specific pumping speed of turbopumps.

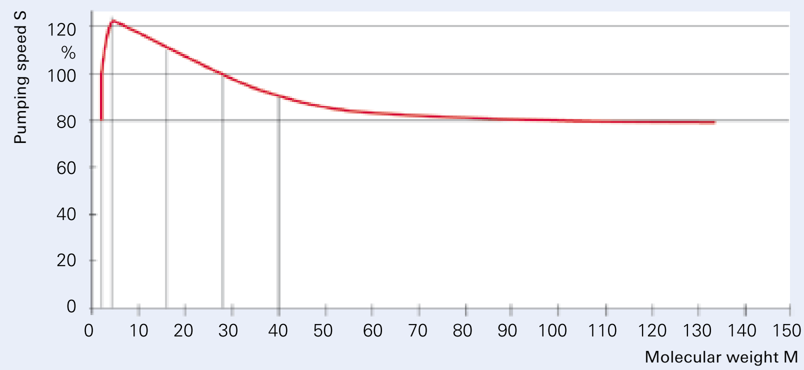
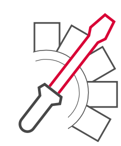


Figure 4: Pumping speed as a function of relative molecular weight.

Points entered in Figure 3 are determined from measured values of the Pfeiffer Vacuum pumps indicated. Points far above the curve are not physically possible. The pumping speeds determined in this manner do not yet say anything about the values for light gases, e.g. for hydrogen (Figure 4). If a turbopump is designed for low ultimate pressure, pump stages with different blade angles are used and the gradation is optimized for maximum pumping speed for hydrogen. This gives pumps with sufficient compression ratios for both hydrogen (about 1,000) and nitrogen. Because of the high partial pressure of nitrogen in air, the compression ratio should be around 10^9. For pure turbopumps consisting of rotor and stator disks, fore-vacuum pressures of about 10^{-2} hPa are required due to molecular flow (Figure 5).



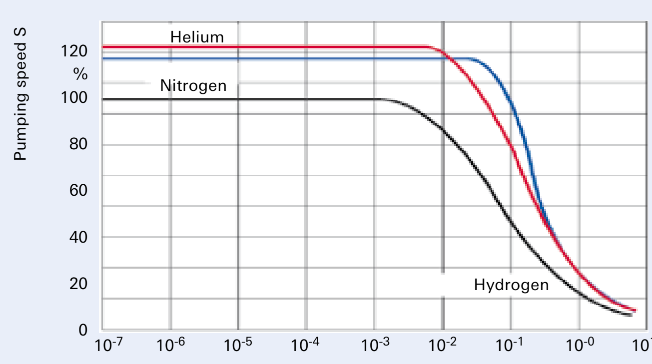
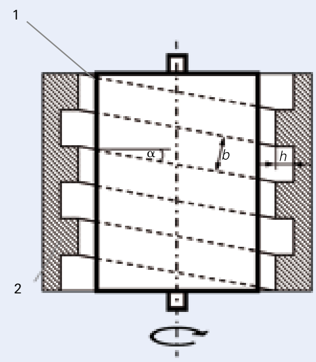


Figure 5: Pumping speed as a function of suction pressure.



1) Rotor
2) Stator with threads

Figure 6: Operating principle of the Holweck stage.

Special features of the Holweck stage

The Holweck stage (Figure 6) is a multistage Gaede molecular pump with a helically wound pump channel. Gas molecules entering the pump channel receive a velocity in the preferential direction of the channel due to the rotation of the rotor. Backflow losses occur due to gaps between the rotor and the webs separating the Holweck channels. To minimize backflow, the gap widths must be kept small.

Cylindrical sleeves (1) are used as the rotor of a Holweck stage, which rotate in helical channels in the stator (2). If stators are arranged both outside and inside the rotor, two Holweck stages can easily be integrated into one pump. In this way, the pumped gas particles are first transported through the stator channels on the outside of the rotor and then back up through further stator channels on the inside of the rotor. From there, they pass through a collecting channel to the backing pump. Modern turbopumps sometimes have several of these "folded" Holweck stages, whose pumping speed S_0 is the same:

$$S_0 = \frac{1}{2} \cdot b \cdot h \cdot v \cdot \cos \alpha$$

Here, $b - h$ is the channel cross-section and $v - \cos \alpha$ is the velocity component in the channel direction. With the channel length L and the velocity $v - \cos \alpha$

$$K_0 = \frac{v \cdot \cos \alpha \cdot L}{\bar{c} \cdot g \cdot h} \quad mit\, 1 < g < 3$$

the compression ratio increases.

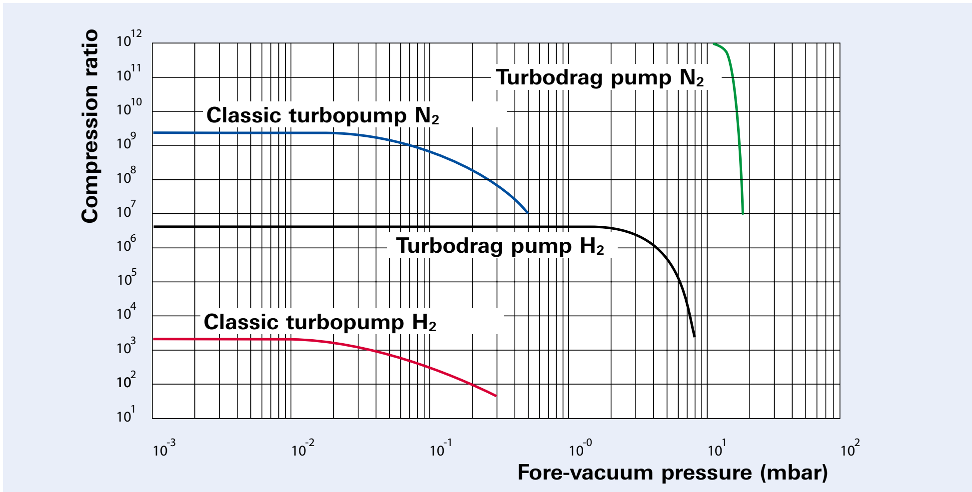


Figure 7: Compression ratios of pure turbo and turbodrag pumps.

Today, turbopumps are equipped with Holweck stages to build up a turbopump ratio with diaphragm pumps and a final pressure between 0.5 and 5 hPa. These are called "turbodrag pumps." Because of the high precompression of the turbopump, only small pumping speeds are needed to generate low base pressures for the Holweck stages. As a result, the delivery channels – especially the channel height and the distances to the rotors – can be kept very small and the molecular flow can be maintained down to the 1 hPa range. The compression ratio for nitrogen is simultaneously increased by the required factor 10^3.

In Figure 9, one can see the shift in the compression ratio curves by about two powers of ten toward higher pressure. A compromise between gas throughput, fore-vacuum compatibility and particle tolerance is made in turbopumps designed for high gas throughput. The gap distances in the Holweck stages are dimensioned somewhat larger in this case.

Figure 8: HiCube Eco turbopump with diaphragm pump

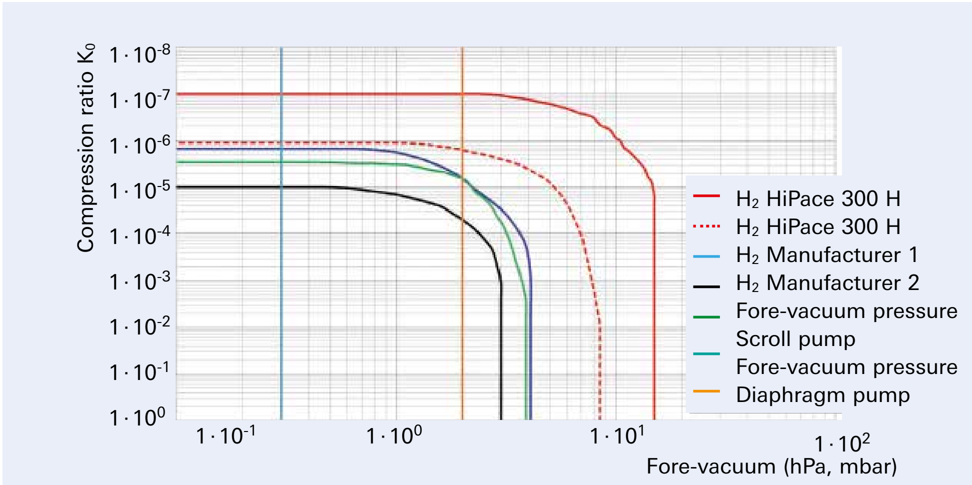


Figure 9: Compression ratios of pure turbo and turbodrag pumps for hydrogen.

Selecting the right backing pump

The compression of the turbopump and backing pump plays an important role in being able to penetrate the lowest pressure ranges. This is particularly true for light gases such as hydrogen. In previous UHV applications, the backing pump already had to provide a low pressure in the order of 10^{-2} hPa. The compression ratio of the turbopump can be based on this. Backing pumps such as rotary vane pumps, multistage Roots pumps or pumping stations can provide such a low backing pressure. Although rotary vane pumps are the less expensive option, there is a risk of oil backflow when the turbopump is switched off, especially in the event of incorrect operation. Dry backing pumps or even pumping stations, which generate a very low backing vacuum, are significantly more expensive and require a relatively large amount of space, which is a disadvantage in many applications. The ideal solution here would be to use a small, low-cost dry backing pump.

Figure 10: Backing pumps – DuoLine rotary vane pump and ACP multistage Roots pump.

Most turbopumps are all-rounders. In addition to good compression, they also offer a large pumping speed and a high gas throughput. However, in very few UHV applications does the high gas throughput play any role at all. Instead, pumping speed and an excellent compression ratio for light gases are what count here. A turbopump's Holweck range optimized for maximum compression values inevitably also reduces the pump's gas throughput. However, this is of secondary importance for the applications mentioned. However, the fact that a large part of the total compression ratio of the backing pump and the turbopump can be transferred to the turbopump is of great advantage. A turbopump with a compression-optimized Holweck stage can therefore discharge against significantly higher backing pressures to achieve the same ultimate pressure. Accordingly, a small diaphragm pump is sufficient to generate ultrahigh vacuum when using turbopumps with a compression-optimized Holweck stage (see Fig. 9, table 1).

Backing pump	**Type**	**Fore vacuum pressure (hPa)**	**Final pressure (hPa)**
two-stage rotary vane pumpe	Duo 6	0.003	$2.47 \cdot 10^{-11}$
Diaphragm pump (2 stage)	MVP 015-2	3.000	$2.75 \cdot 10^{-11}$
Diaphragm pump (3 stage)	MVP 020-3	1.300	$2.65 \cdot 10^{-11}$
Multi-stage Roots pump	ACP 15	0.032	$2.56 \cdot 10^{-11}$

Table 1: Ultimate pressure as a function of various backing pumps generated with a HiPace 300 H.

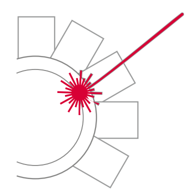
Figure 11: HiPace 300 H turbopump

Such optimized turbopumps have a high fore-vacuum compatibility, so that the diaphragm pump can unquestionably still be operated in interval mode. It only has to be switched on if the backing vacuum pressure reaches an impermissibly high value. Numerous applications have shown that the diaphragm pump is in operation less than 10% of the total time. In addition to the resulting energy savings, the lower heat radiation of the backing pump and also the ultimately virtually noiseless operation in laboratories should not be underestimated.

Moreover, in order to maintain an extremely low pressure (see Figure 9 and Table 1), ion getter pumps, which are normally connected downstream of the turbopump, are no longer necessary.

Thus, with an intelligent interconnection of Holweck stages in modern turbopumps, compression can be significantly increased, especially for light gases. Simple, small backing pumps can be used to generate very low pressures in the low UHV range. Compared to the options used in the past, this is a very big advantage. However, it is also important to point out the limitations of these solutions: A high-compression turbopump is less suitable for large gas loads.

In 2021, Pfeiffer Vacuum introduced laser balancing technology. This is a novel method that provides even more efficient balancing of the rotors of turbopumps. Instead of screwing balancing weights into balancing holes, material is removed with the aid of a high-energy laser beam. Mass balancing is thus reversed. This method ensures significantly lower residual imbalance, thus reducing vibrations and noise emissions and extending the service life of the ball bearing. Pfeiffer Vacuum's turbopumps can thus be used even more efficiently. The technology is patented and unique on the market worldwide.



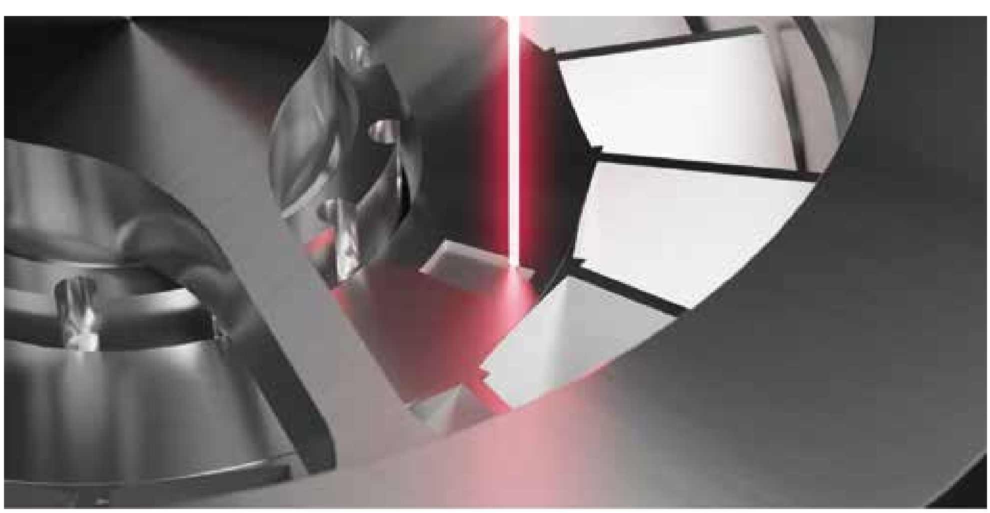


Figure 12: As a pioneer in laser ablation, Pfeiffer Vacuum is able to balance the rotor in the nanogram range

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